Comparison on Thermodynamic Performances between Basic and Regenerative Sub-Critical Organic Rankine Cycles Coupled with Low Grade Heat Source

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Comparison on Thermodynamic Performances between Basic and Regenerative Sub-Critical Organic Rankine Cycles Coupled with Low Grade Heat Source

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Abstract. The coupling effects of low grade heat source on thermodynamic performances of both the basic and regenerative sub-critical Organic Rankine cycle (ORC) systems are investigated and compared. Turbine inlet pressure and turbine inlet temperature are treated as independent variables. The system exergy efficiency is selected as the thermodynamic performance criterion. The waste hot water with temperature range of 373.15K to 423.15K is selected as the low grade heat source. Optimization studies are carried out by using eight common organic working fluids to recover heat energy, and relevant operating conditions are obtained respectively. The results indicate that both hot fluid inlet temperature and the allowable minimum hot fluid outlet temperature influence the optimal working fluid, relevant turbine inlet condition and system exergy efficiency. Optimal system exergy efficiency increases monotonously with increasing of hot fluid temperature for both ORC systems above. In comparison to the basic cycle configuration, optimal system exergy efficiency for the regenerative system using the same working fluid is not changed if relevant turbine inlet vapor is saturated, but is significantly improved if relevant turbine inlet vapor is overheating. Besides, the optimal working fluid is dependent on the heat source temperature and specific cycle configuration.

1. Introduction

In the past few decades, organic Rnakine cycle (ORC) technology has drawn an increasing attention to harvest energy from the low grade heat sources. Generally, the low grade heat sources refer to the heat sources with temperature less than 473K, including the industrial waste hot water, the exhaust gas from internal combustion engines and gas turbines, the geothermal brine, the solar hot water, etc. The working principle of the ORC system is almost identical with the conventional steam Rankine system, except for the usage of alternative working fluid. In accordance with the lower temperature of heat sources, organic fluid with much lower critical temperature than water is selected as the working fluid.

For a new ORC power plant design, the first issue is to define the cycle configuration. Many cycle configurations such as sub-critical, trans-critical or super-critical, basic or regenerative, single-pressure or dual-pressure cycles [1, 2] have been proposed and investigated. In practice, the basic and regenerative, sub-critical and single pressure ORC systems are universally adopted in many state-of-the-art applications due to its allowable working pressure and system compactness [3, 4]. Therefore, the present study is focus on the basic and regenerative sub-critical ORC systems. In comparison to the basic cycle configuration, the use of regenerator can lower the heat load into the evaporator and thus improve cycle thermal efficiency. Relevant studies [5-7] showed that the benefit of using regenerator can be neglected in terms of net power output or specific net power output.

Although the layout of the basic or regenerative sub-critical ORC system is relatively simple, it is not an easy work to obtain the optimal performance and relevant working condition. Many efforts have been made to obtain the optimal working conditions and maximum thermodynamic performance of the ORC system. Yamamoto et al [8] concluded that saturated vapor for organic fluid at turbine inlet gives higher turbine power output than overheating vapor. Based on this conclusion, subsequent optimization works [9, 10] are mostly carried out on the evaporating pressure or temperature along the saturated vapor line. By contrast, Wang et al [11] and Satanphol et al [12] increased the investigation of operating conditions including saturated and superheated states at turbine inlet, and treated the turbine inlet pressure and turbine inlet temperature as independent variables. However, the previous studies [8-12] neglected the coupling effect of low grade heat source, i.e. the heat power recovery capability from the low grade heat source was not considered. The trade-off relation between cycle thermal efficiency and heat source recovery effectiveness was firstly discussed in the work of Liu et al [13]. They concluded that the cycle thermal efficiency alone may be an inappropriate criterion for thermodynamic performance evaluation, and a new criterion named system thermal efficiency was proposed. Borsukiewicz-Gozdur A [14] introduced two concepts about open and sealed type of heat sources, which highlight the difference between heat source capacity and heat power supplied to the cycle, the power plant exergy efficiency was finally proposed as a new criterion for the open type heat source recovery. This new performance evaluation criterion in the view of whole system is adopted by subsequent researchers [15, 16].

The aim of present paper is to perform parametric optimization and performance comparison between the basic and regenerative sub-critical ORC systems by using several common working fluids, especially considering the coupling effect of the low grade heat source.

2. ORC system modelling

2.1. ORC system description

The schematic layout of the regenerative ORC system is shown in figure 1. This ORC system consists of five components: an evaporator, a turbine, a regenerator, a condenser and a feed pump.

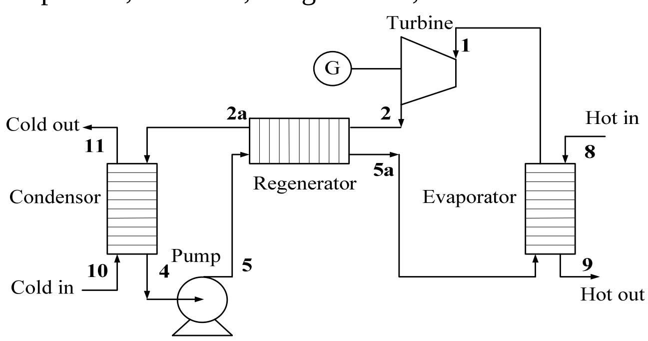

Figure 1. .The layout of regenerative ORC system

Figure 2 shows the thermodynamic processes in terms of temperature-entropy diagrams of the basic and regenerative ORC systems. The corresponding thermodynamic processes of working fluid in the basic cycle configuration are: process 1-2 (expansion process across the turbine), process 2-4 (heat removal by the cold fluid at constant low pressure), process 4-5 (pumping process), and process 5-1 (heat addition from the hot fluid at constant high pressure). Moreover, process 8-9 accommodated with process 5-1 in evaporator depicts the decreasing trend of hot fluid temperature, and process 10-11 accommodated with process 2-4 in condenser describes the increasing trend of cold fluid temperature. For the regenerative cycle configuration, the working processes of the working fluid are changed to process 5a-1 in the evaporator and 2a-4 in the condenser, respectively. In addition, evaporator pinch point temperature difference $\Delta T_{pp,e}$, condenser pinch point temperature difference $\Delta T_{pp,c}$, and regenerator pinch point temperature difference $\Delta T_{pp,r}$ are depicted in figure 2.

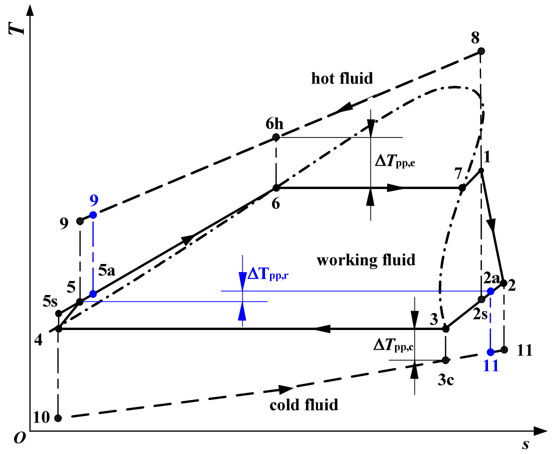

Figure 2. *T-s* diagrams for the basic and regenerative ORC systems. The blue lines indicate the regenerative cycle configuration.

2.2. Mathematical modelling

Based on the energy conservation under steady operating condition, the work output by turbine is given by

$$W_{\mathrm{t}} = m_{\mathrm{wf}}\left(h_1 - h_{2\mathrm{s}}\right)\eta_{\mathrm{t}} \tag{1}$$

Where, W_t is the work output by turbine (W), m_{wf} is the mass flow rate of working fluid (kg/s), h_1,h_{2s} are the specific enthalpies at states 1 and 2s (J/kg), and η_t is the turbine isentropic efficiency.

The work consumption by the feed pump can be calculated as

$$W_{\mathrm{p}} = m_{\mathrm{wf}}\left(h_{5\mathrm{s}} - h_4\right)/\eta_{\mathrm{p}} \tag{2}$$

Where, W_p is the work consumption by the feed pump(W), h_{5s},h_4 are the specific enthalpies at states 5s and 4 respectively (J/kg), and η_p is the feed pump isentropic efficiency.

The low grade heat source recovery is usually constrained by the physical characteristics of hot fluid, hence the hot fluid outlet temperature is restricted by given minimum allowable value T_{9min}. Hence, system exergy efficiency, which is used to judge the overall utilization degree of the heat source, can be calculated as

$$\eta_{\mathrm{sym,ex}} = \frac{W_{\mathrm{t}} - W_{\mathrm{p}}}{m_{\mathrm{hf}}\left[\left(h_8 - h_{9\min}\right) - T_0\left(s_8 - s_{9\min}\right)\right]} \tag{3}$$

Where, $\eta_{sym.ex}$ is the system exergy efficiency, m_{hf} is the mass flow rate of hot fluid (kg/s), h_8,h_{9min} are the specific enthalpies at state 8 and with minimum allowable hot fluid outlet temperature (J/kg), s_8, s_{9min} are the specific entropies at state 8 and with minimum allowable hot fluid outlet temperature (J/kg·K), and T_0 is the reference temperature with value of 293.15K.

3. ORC system specifications

3.1. Specified conditions

The low grade heat source is waste hot water with the mass flow rate of hot water is 10kg/s. The minimum allowable hot water outlet temperature is set as 333.15K. The heat sink is cold water at 293.15K. The isentropic efficiencies of turbine and feed pump are set as 70% and 75% respectively. The pinch point temperature differences in evaporator, condenser and regenerator are 10K, 5K and 5K respectively. The condensing temperature is set as a typical value of 303.15K. All the above mentioned parameters are given with fixed values. Furthermore, the temperature of hot fluid varies from 373.15K to 423.15K to investigate the influence of heat source temperature.

The independent variables are turbine inlet pressure and turbine inlet temperature. Following the suggestion in [17], the valid working regions of independent variables (i.e. turbine inlet pressure and temperature) are given in the following:

$$p_{\mathrm{cond}} + 10\mathrm{kPa} \le p_1 \le p_{\mathrm{sta}}(\min\lfloor(T_{\mathrm{crit}} - 10\mathrm{K}), T_8 - \Delta T_{\mathrm{pp,e}}\rfloor) \tag{4}$$

$$T_{\mathrm{sta}}(p_1) \le T_1 \le T_8 - \Delta T_{\mathrm{pp,e}} \tag{5}$$

Besides, expansion process 1-2 across the turbine is constrained to locate at the superheated region, i.e. the dryness fraction of working fluid in process 1-2 is always higher than or equal to be 1.

3.2. Candidate working fluids
Based on the existing criteria for working fluid selection in [18-20], eight candidate working fluids R123, R245fa, R114, R236ea. R236fa, RC318, R227ea and R1234yf in present study are all dry or isentropic fluids. The main thermodynamic properties of the selected fluids are summarized in Table 1.T_{crit} and p_{crit} denote the critical temperature (K) and critical pressure (kPa) of the working fluid. All the thermodynamic properties of candidate working fluids are acquired from NIST REFPROP database. It is nothing that the condensing pressures are all higher than the atmosphere pressure to avoid the utilization of vacuum facilities for keeping vacuum conditions in condenser and turbine.

Table 1. Thermodynamic properties of candidate working fluids

Working Fluid	T_{crit}(K)	p_{crit}(kPa)	p_{cond}^{a}(kPa)
R123	456.83	3661.80	130.53
R245fa	427.16	3651.00	211.73
R114	418.83	3275.00	292.25
R236ea	412.44	3502.00	288.12
R236fa	398.07	3200.00	375.96
RC318	388.38	2777.50	425.15
R227ea	374.90	2925.00	610.79
R1234yf	367.85	3882.20	895.21

[a] Condensing pressure at T_3=303.15K.

4. Results and discussion

4.1. Optimal working condition
Figure 3 shows the optimal turbine inlet conditions with maximum system exergy efficiency using working fluids R114, R236fa and R1234yf with three heat source temperatures T_8=373.15K, 393.15K and 423.15K. It is noticed that all the optimal turbine inlet conditions always lie on the saturated lines for working fluids R114 and R236fa with either heat source temperature. But using working fluid R1234yf, the optimal turbine inlet conditions do not always locate on the saturated line. The optimal turbine inlet condition moves to the overheating region with the increasing of heat source inlet temperature. Besides, the optimal turbine inlet pressure and temperature increase with the increasing

of heat source temperature for either specific working fluid. The regenerator has no effect on the optimal working conditions for working fluids R114 and R236fa, and influences the optimal working conditions for working fluid R1234yf.

4.2. Optimal system exergy efficiency

The change trends of optimal system exergy efficiency versus heat source temperature using working fluids R114, R236fa and R1234yf are shown in Figure 4. When optimal turbine inlet temperatures are kept at saturated states (see figure 3) using working fluids R114, R236fa, and R1234yf, optimal system exergy efficiencies monotonously increase with the increase of heat source temperature for either cycle configuration. The regenerative cycle using working fluid R1234yf also has increasing system exergy efficiency with the increasing of heat source temperature. But the basic cycle configuration using working fluid R1234yf shows a local maximum system exergy efficiency with heat source temperature T_8=403.15K. Compared with the system exergy efficiency for the basic cycle configuration, the system exergy efficiencies for the regenerative cycle configuration using working fluids R114 and R236fa show the same values. The regenerative cycle configuration exhibits more or less benefits when the optimal turbine inlet conditions are overheated (see figure 3) with heat source temperature T_8=403.15K - 423.15K.

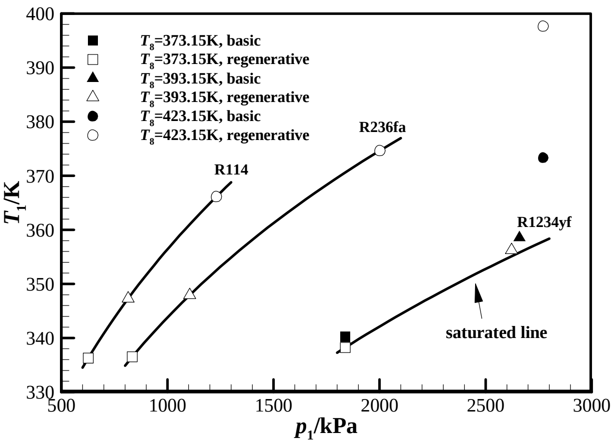

Figure 3. Optimal turbine inlet conditions for the basic and regenerative ORC systems. The solid lines indicate saturated lines for working fluids R114, R236fa and R1234yf.

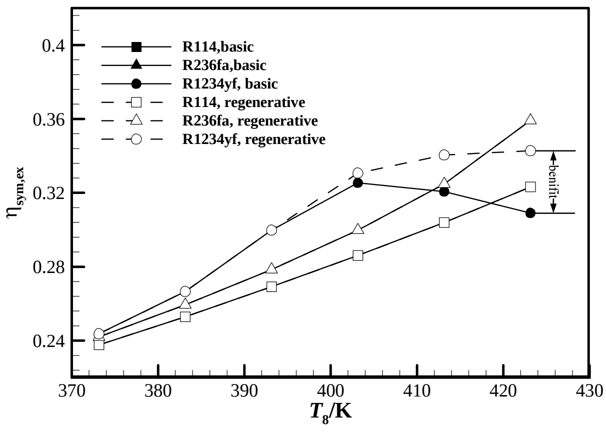

Figure 4. Optimal system exergy efficiency versus heat source temperature for the basic and regenerative ORC systems

The system exergy efficiencies for the basic and regenerative ORC system, and the benefits of regenerator using working fluids RC318, R227ea and R1234yf are listed in table 2. The maximum benefit of system exergy efficiency is 5.44% for working fluid RC318 with T_8=423.15K, and the minimum value is 0.53% for working fluid R1234yf with T_8=403.15K. The benefit is bigger with higher heat source temperature for a specific working fluid. With a certain heat source temperature, benefit of the regenerative ORC system decreases with the decreasing of critical point temperature of working fluid.

Table2. System exergy efficiency benefits of the regenerative ORC system.

Working fluid	T_8=403.15K			T_8=413.15K			T_8=423.15K		
	basic	regenerative	benefit	basic	regenerative	benefit	basic	regenerative	benefit
RC318							0.3509	0.4053	0.0544
R227ea	0.3401	0.3490	0.0089	0.3324	0.3701	0.0377	0.3173	0.3629	0.0456
R1234yf	0.3254	0.3307	0.0053	0.3207	0.3405	0.0198	0.3091	0.3428	0.0337

The working fluids with maximization of system exergy efficiency with different heat source temperatures for the basic and regenerative ORC systems are shown in figure 5. Note that the BSUB and RSUB are the abbreviations of the basic and regenerative sub-critical ORC systems, respectively. With the heat source temperature T8=373.15K-403.15K, best performances are obtained with R227ea as working fluid, and there are no differences of the system exergy efficiencies between the basic and regenerative ORC systems. System exergy efficiencies for the regenerative ORC system are larger than the basic ORC system with heat source temperature of T_8=403.15K-423.15K. With heat source temperature of T_8=413.15K, optimal system exergy efficiency is achieved with value of 35.41% for working fluid RC318 for the basic ORC system and 36.98% for working fluid R227ea for the regenerative ORC system. The maximum system exergy efficiency is 40.53% for working fluid RC318 for the regenertive ORC system, and 35.92% for working fluid R236fa for the basic ORC system with the hot fluid inlet temperature of T_8=423.15K.

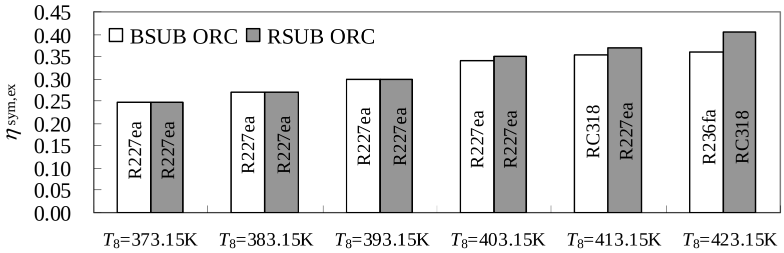

Figure 5. Best performance working fluids and relevant system exergy efficiencies.

5. Conclusion

Thermodynamic performance analysis and parameter optimization of the basic and regenerative ORC systems are performed considering the coupling effect of low grade heat source. The following conclusions can be drawn from the results of the presented work:

(1) Under the given conditions, whether saturated or overheated state for optimal turbine inlet condition is influenced by the heat source temperature, the critical temperature of working fluid and the limitation of minimum allowable heat source outlet temperature. If optimal turbine inlet vapor is saturated, optimal system exergy efficiency of the regenerative ORC system using the same working fluid is kept the same value as the one of the basic ORC system. While optimal turbine inlet vapor is located in the overheating region, optimal system exergy efficiency of the regenerative ORC system is higher.

(2) For a specific working fluid, the benefit is bigger with a higher heat source temperature. The benefit of the regenerative ORC system decreases with the decreasing of critical point temperature of working fluid when the heat source temperature is certain.

(3) Working fluid R227ea always shows best performances for both the basic and regenerative ORC systems with heat source temperature in range of 373.15K to 403.15K. The best performance working fluid is RC318 for the basic ORC system with hot fluid inlet temperature of 413.15K and for the regenerative ORC system with heat source temperature of 423.15K. Working fluids R227ea and R236fa give the highest system exergy efficiency for the regenerative ORC system with heat source temperature of 413.15K and for the basic ORC system with heat source temperature of 423.15K respectively.

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